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(54) Title: A VEHICLE SUSPENSION CONTROL SYSTEM

(57) Abstract

The invention provides a vehicle suspension control system comprising means for determining forces acting between the sprung mass (body) of a vehicle and one or more unsprung masses (wheel and hub assemblies) connected thereto and producing signals proportional to the forces, means for resolving the force signals to represent steady state and dynamic force components respectively, first means for determining from the steady state force signals values of vector quantities required between the sprung mass and the or each unsprung mass to maintain a constant orientation of the sprung mass relative to a reference plane, second means for determining, from the dynamic force signals, values of vector quantities required between the sprung mass and the or each unsprung mass substantially to eliminate the transmission of dynamic forces between the sprung mass and the or each unsprung mass and also means for applying the required vector quantities. The invention relates to a control system which analyses the forces experienced by a vehicle suspension system in two categories, dynamic forces and steady state forces. The dynamic forces experienced by the vehicle are those occasioned by road inputs and the steady state forces experienced by the vehicle are those occasioned by vehicle motion.
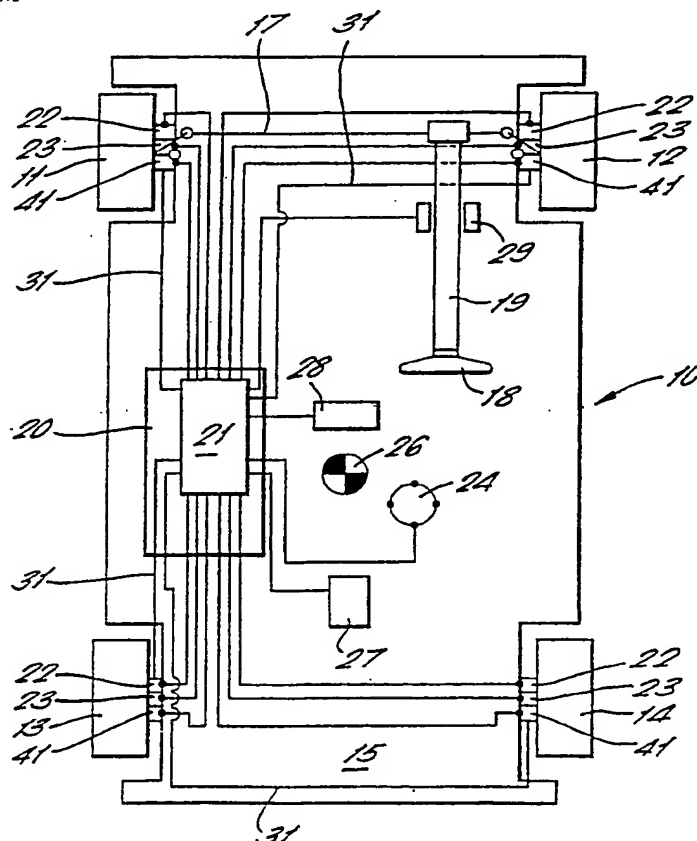
DESIGNATIONS OF "DE"

Until further notice, any designation of "DE" in any international application whose international filing date is prior to October 3, 1990, shall have effect in the territory of the Federal Republic of Germany with the exception of the territory of the former German Democratic Republic.

A VEHICLE SUSPENSION CONTROL SYSTEM

This invention relates to a vehicle suspension
control system. By "vehicle" is meant all classes of
vehicle capable of motion on land, and the term
includes motor vehicles, lorry cabs and trailers,
tractors and tracked vehicles.

In particular, the invention relates to a
vehicle suspension control system for controlling the
suspension of a vehicle having an active suspension
system.

Active suspension systems are well known. For
example, our European Patent Application No.
EP-A-0114757 discloses an active suspension system in
which a number of sensor inputs representing
variables of vehicle motion are processed in a
microprocessor to produce signals representing forces
required at the points of support of the vehicle body
on its wheel/hub assemblies. The required forces are
applied by respective hydraulic actuators disposed
between the vehicle body and each wheel/hub assembly
to maintain as nearly as possible a constant attitude
of the vehicle.

Active suspension systems operate on the
principles either of measuring the forces acting
between the vehicle body and the wheel/hub assemblies
or of calculating such forces from measured values
indicative of vehicle behaviour.

The main problem faced by the designer of an
active suspension system is the difficulty that
whilst the vehicle suspension system should respond
to perturbations in the road (dynamic inputs) by
moving the actuators attached to the wheel and hub
assembly, the suspension system should not cause the
actuators to deflect in response to loads imposed

upon the vehicle by reason of "steady state inputs".
"Steady state inputs" include loads being placed on
the suspension due to the cornering and
acceleration/deacceleration of the vehicle. "Steady
5   state input" also include variations on the load of
the vehicle due to load carried inside the vehicle
and aerodynamic forces on the vehicle. Such "steady
state inputs" should generally not cause displacement
of the actuators. In other words, the designer is
10   faced with the problem of designing a suspension
system which is "soft" to road input whilst being
"hard" to steady state inputs.

According to the invention, there is provided a
vehicle suspension control system comprising:
15   means for determining forces acting between the
sprung mass of said vehicle and one or more unsprung
masses connected thereto and producing signals
proportional to said forces;
means for resolving said force signals to
20   represent steady state and dynamic force components
respectively;
first means for determining, from the steady
state force signals, values of vector quantities
required between the sprung mass and the or each
25   unsprung mass to maintain a constant orientation of
said sprung mass relative to a reference plane;
second means for determining, from the dynamic
force signals, values of vector quantities required
between the sprung mass and the or each unsprung mass
30   substantially to eliminate the transmission of
dynamic forces between the sprung mass and the or
each unsprung mass; and
means for applying said required vector
quantites.
35   There now follows a description of a specific
embodiment of the invention, by way of example, with

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referernce being made to the accompanying drawings in
which:

Figure 1 is a schematic plan representation of
a motor vehicle having a control system according to
5   the invention;

Figure 2 is a schematic elevational view of one
corner of the vehicle of Figure 1; and

Figure 3 is a block diagram showing the
principles of operation of the control system of
10   Figures 1 and 2.

Referring to Figure 1 there is shown a motor
vehicle 10 having four wheels 11, 12, 13, 14. The
motor vehicle 10 has a body 15 and the wheels 11, 12,
13, 14 are connected to the body 15 by suspension and
15   steering or drive components (not shown in detail).
The suspension components are described below in
relation to Figure 2.

The front wheels 11 and 12 of the vehicle 10
are steerable, and the vehicle has a conventional
20   steering linkage 17 driven by steering wheel 18 via
steering column 19.

The vehicle 10 of Figure 1 has an on-board
computer 20 incorporating a microprocessor 21.
Microprocessor 21 receives signals from a number of
25   sensors distributed about the vehicle and controls
the suspension of the vehicle 10 in response to the
signals.

Each wheel and its associated hub assembly has
three transducers capable of sending signals to
30   microprocessor 21, these transducers being load cells
22, accelerometers 23 and displacement transducers.

The load cells 22 respectively measure the load
acting between the top of each suspension strut 30
(best shown in Figure 2) and the vehicle body 15.
35   The accelerometers 23 measure the respective vertical
accelerations of the wheel/hub assemblies.

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The forces transmitted via the load cells 22
and calculated from the accelerometer measurements
are the dynamic forces acting on the vehicle body 15,
and are occasioned, for example by the vehicle 10
5   encountering variations in the road surface along
which it travels. The load cells 22, accelerometers
23 and microprocessor 21 therefore constitute means
for determining forces acting between the sprung mass
(ie body 15) of the vehicle and one or more unsprung
10   masses (ie the wheel/hub assemblies) and producing
signals proportional to the forces. The load cells
22, accelerometers 23 and microprocessor 21 also
constitute means for resolving the dynamic force
components.
15   The microprocessor 21 also receives signals
from further transducers which signals are processed
in the microprocessor 21 to represent steady state
forces acting between the vehicle body 15 and each
wheel/hub assembly.
20   A gyrometer 24 located near the vehicle centre
of gravity indicated by reference numeral 26 measures
the yaw rate (ie rotational velocity) of the vehicle
10. Signals are generated by the gyrometer 24 when,
for example, the vehicle follows a curved path.
25   A longitudinal accelerometer 27 indicates, for
example, the extent to which the vehicle is
accelerating and braking.
        A lateral accelerometer 28 measures the vehicle
lateral acceleration. The lateral accelerometer is
30   also adjacent the vehicle centre of gravity 26 and
indicates, for example, the lateral acceleration
effects of the vehicle travelling around a curve, and
of the lateral component of gravitational
acceleration when the vehicle 10 is on a cambered
35   road.
        A steering angle sensor 29 located in the

embodiment of Figure 1 on the steering column 19
measures the angle β through which the steering
wheel is turned. The steering angle sensor could be
located in one of a variety of positions, for

5   instance a suitable transducer could be included in
the steering linkage 17 or on one of the front
wheel/hub assemblies.

The transducers 24, 27, 28 and 29 in
combination with microprocessor 21 constitute further

10   means for determining forces acting between the
sprung mass (ie the vehicle body 15) of the vehicle
10 and one or more unsprung masses (ie the wheel/hub
assemblies) and producing signals proportional to the
forces; and also constitute means for resolving the

15   steady state components of the forces.

The steady state forces acting between the
vehicle body 15 and the wheel/hub assemblies are
therefore those occasioned by the condition of the
vehicle as determined by the driver thereof, for

20   example whether the vehicle is decelerating being one
such condition.

The microprocessor 21 is capable of
determining, from the steady state force values,
vector quantities in the form of velocities required

25   between the sprung mass (ie the vehicle body 15) and
the unsprung masses (ie the wheel/hub assemblies) to
maintain a constant orientation of the vehicle body
relative to a predetermined reference plane.

The microprocessor 21 is also capable of

30   determining from the dynamic force values the values
of vector quantities in the form of velocities
required between the body 15 and the wheel/hub
assemblies to eliminate the transmission of dynamic
forces therebetween.

35   The required velocities are transmitted as
electrical signals to the wheel/hub assemblies via

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output lines 31 where actuators 32 incorporated
within the suspension struts operate to apply the
required velocities between each wheel/hub assembly
and the vehicle body 15.

5   Figure 2 shows schematically the arrangement of
components forming the suspension strut associated
with one corner of the vehicle 10. Also shown in
Figure 2 is the arrangement of the pump 33 and
hydraulic supply circuit for the actuator 32 shown
10   therein. Clearly, the circuit shown in Figure 2
represents one quarter of the circuitry for the
entire vehicle 10, and whilst the pump 33, hydraulic
control circuit 32 and microprocessor 21 are common
to all four suspension assemblies of the vehicle, the
15   electrical control lines between the suspension
assembly and the microprocessor 21; the hydraulic
supply and return lines 36 and 37; and the suspension
strut 30 are reproduced for the three suspension
assemblies not shown in Figure 2.

20   In Figure 2, the sprung mass of the vehicle in
the form of the vehicle body 15 is shown supported on
a suspension strut indicated generally by the
reference sign 30, which are in turn supported on a
wheel and tyre modelled as an unsprung mass 35 in the
25   form of the wheel/hub assembly supported on a spring
36 and damper 37 representing the tyre
characteristics.

The suspension components 30 comprise means for
applying vector quantities in the form of velocities
30   to control the attitude of the vehicle, in the form
of an hydraulic actuator 32, shown vertically aligned
and secured at its upper end to a load cell 22 which
is separated from the vehicle body 15 by an isolator
38, which may be, for example, a rubber block. The
35   actuator 32 need not necessarily be vertically
alinged, depending on the space available for

suspension components and the suspension layout
adopted.   The load cell 22 is capable of measuring at
least a portion of the loads acting between the
wheel/hub assembly and the vehicle body and producing
5   signals proportional to the loads.

A spring 39 is shown connected in parallel with
the hydraulic actuator 32.

The spring 39 does not control the attitude of
the vehicle in the way that it would in a vehicle
10   having a conventional suspension system.   Spring 39
serves merely to reduce the power consumption of the
control system of the invention by bearing a
significant proportion of the static load of the
vehicle body 15.

15   Thus, the operation of actuator 32 may take
place over a wide range of displacements actually to
effect control of the vehicle without requiring an
excessive power consumption as would normally be
required if the actuator were to support the static
20   load of the vehicle body 15 in addition to
controlling the steady state and dynamic loadings
resulting from dynamic and steady state forces acting
on the vehicle.

The actuator 32 is preferably a double-acting
25   electro-hydraulic actuator comprising a sealed
cylinder, a piston attached to a connecting rod and a
flow control valve.   The sealed cylinder is capable
of containing high pressure fluids without
significant leakage.   The piston is attached to a
30   connecting rod and is housed within the cylinder.
The piston device is cylindered in to two chambers.
The piston is fitted with seals to prevent
significant leakage from one chamber to another.

The hydraulic circuit 34 preferably comprises a
35   four port control valve, or electro-hydraulic servo
valve (EHSV), which has two input and two output

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ports. One input port is connected to a source of
high pressure hydraulic fluid, the other input port
is connected to a return path which is held at low
pressure. The EHSV is designed such that when an
5   electric current is passed through its energising
coil, hydraulic fluid is allowed to flow from the
high pressure source in to one cylinder chamber of
the actuator 32, whilst fluid is allowed to flow from
the other chamber of the actuator 32 in to the return
10   path. This causes the piston to move relative to the
cylinder with a velocity proportional to the electric
current applied to the energising coil and the load
reacted by the differential pressure across the
piston. When the plurality of the current is
15   reversed the velocity of the piston is reversed. If
the EHSV conforms to the ideal, then the velocity of
the piston will be zero when the electrical current
applied to the energising coil is zero.

Since the power consumption of actuator 32 is
20   reduced, by the use of spring 39, its piston area may
be designed to be relatively small, thereby producing
a compact device. Further, spring 39 serves as a
fail-safe device in that it supports the static load
of the vehicle body 15 in the event of total failure
25   of the control system of the invention.

The input and output ports of the hydraulic
actuator 32 are connected via hydraulic pipework 32a
and 32b to a hydraulic control circuit 34 including a
suitable supply pump 33. The hydraulic circuit 34
30   operates, via electrical connection 34', under the
command of microprocessor 21 which produces a
demanded output of the actuator 32 in response to the
number of measured inputs.

The suspension strut 30 and hydraulic circuit
35   34 therefore constitute means for applying required
vector quantities between the vehicle body 15 and the

wheel/hub assembly in the form of unsprung mass 35.

Electrical signals monitoring the hydraulic system pressure and, where appropriate, the pump swash plate angle may optionally be fed into
5   microprocessor 21 via lines 34" and 33'. Further, the position of actuator 32 is measured by Linear Variable Inductive Transducer (LVIT) 41 and fed via line 41' to microprocessor 21.

Clearly, alternatives to the kinds of
10   transducer, pump and actuator described herein may be used in the embodiment of Figures 1 and 2 if required.

Turning now to Figure 3, there is shown a schematic block diagram of the manner in which the microprocessor 21 conditions the various signals to
15   produce the output signals transmitted via line 34'.

In Figure 3, a chain line divides the diagram into three portions. Portion A, occuring generally above the chain line, represents those steps
20   concerned with dynamic forces acting on the vehicle; portion B, occuring generally below the chain line represents processing steps concerned with steady state forces acting on the vehicle; and portion C represents the final output of the microprocessor,
25   this being a combination of the output of portions A and B.

The signals from the four load cells 22 and hub accelerometers 23 are fed into block 42 which reconfigures the dynamic force values represented by
30   the signals as modal forces acting on the vehicle body, the modes corresponding to the four rigid body modes of displacement, ie the heave, pitch, roll and warp modes. The forces are calculated as being forces at the 'tyre patch', that is to say the point
35   of contact of the tyre with the road. These forces are calculated as being the sum of the measured

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forces and the product of the masses of the unsprung masses with their respective accelerations. This sum is necessary since force in a tyre at the tyre patch will not all be transmitted to the body of the
5   vehicle; a portion will accelerate the wheen and hub assembly to which the tyre is attached.

In Block 43 unmeasured modal forces are added to the measured modal forces calculated in Block 42. The applicant envisages a situation wherein springs
10   are used in parallel with the actuators in the vehicle suspension and only the force transmitted by the actuators to the body is measured. The force transmitted to the body by the springs can be calculated from measured displacements of the
15   actuator, which are fed to the box 42 from the LVITs 41. The modal loads may also be adjusted to compensate for unmeasured loads, transmitted to the vehicle body by the suspension linkages and also components of horizontal load on the wheels
20   transmitted through the actuator by reason of imperfection suspension arrangements.

The Block 44 removes from the modal forces calculated those forces due to the steady state inputs to the vehicle, which are calculated in
25   Portion B, discussed below.

The corrected modal vector can be written:

$$\{Fcm\} = (Tfm).((Fr)+(Mm).\{DDXu\})$$

30                   $$+ (\Gamma in).\{\Phi e\}+(\Gamma sp).\{Xdm\}$$

                     $$+ [Kt].\{Xx\}$$

                          where

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$\{Fcm\}$ is the vector of corrected modal forces

(Fr) is the vector of measured loads

(DDXu) is the vector of measured unsprung mass accelerations

(Φe) is the transpose of the vector (nx^+, nx^-, ny, Dr)

where nx is longitudinal acceleration

ny is lateral acceleration

Dr is yaw acceleration

(Xdm) is the vector of modal actuator modal displacements

(Xx) is the position offset vector

(Kt) is a diagonal matrix of tyre/isolator stiffness

(Mm) is a diagonal matrix proportional to hub/wheel masses

(Tfm) is the load modal transformation matrix.

The block 46 sythesises four modal spring and damper units and calculates modal velocity demands. The modal velocity demands are then turned into individual actuator co-ordinates.

The signals from accelerometers 23 are, in addition, processed in block 45 as individual hub velocity demands which are fed forwardly to block 46 for combination with the calculated corner velocity demands. The suspension simulation is further enhanced by the adding of the weighted unsprung mass inertial velocity to each actuator demand. This provides the potential for introducing inertial damping of the unsprung mass.

The output of block 46 is filtered in high pass filter block 47. This is necessary to ensure that measurement bias errors do not corrupt the simulation.

Block 48 represents an electronic bump stop control which may optionally modify the output of portion A of the program. The electronic bump stop

control is disclosed in a co-pending patent
application of the applicants.

The outputs of block 47 and 48 are combined in a
junction 48 and this velocity demand combined output
5 is input to an arrangement 50 which issues electrical
control signals to the actuators. A suitable
arrangement is described in a co-pending application
of the applicant, but the arrangement could be of many
forms well known in the prior art.

10 The output signals representing the steady state
forces, from the LVIT 41, gyrometer 24, longitudinal
accelerometer 27, lateral accelerometer 28, a vehicle
speed sensor 25 and the steering angle sensor 29 are
conditioned in corresponding blocks 41a, 24a, 27a,
15 28a, 25a and 29a to ensure conformity of signal types
(ie to resolve the measured steady state variables
into modal force values).

The outputs of these blocks are fed in to block
52, in which the dynamic forces of heave, pitch, roll
20 and warp are calculated.

The block 53 of the control system models a
series of actuators, controlling the suspension of the
vehicle. The dynamic forces are converted in to
required corner displacements, which are output to the
25 control system 50 which in turn directly controls the
actual actuators of the vehicle.

Portion B instructs the actuators to extend and
contract to improve the vehicle ride and handling.

The additional motion of the actuators demanded
30 by portion B could alter measured loads, which would,
in turn, cause an unwanted reaction from the simulated
suspension in portion A. However, as mentioned
before, the changes in measured loads to be expected
from such additional actuator motion are estimated and
35 subtracted from the measured loads in order to
eliminate the unwanted reaction from the simulated

suspension of portion A.

If the displacement required of the actuators by the simulated suspension of portion A is averaged over a long period of time, then the average displacement

5   should be zero. The portion B acts to control the average position of the actuators in order to improve the ride of the vehicle.

Theoretically, the portion B considers the four actuators connected in series to the four actuators

10   controlled by portion A. Displacement of the theoretical actuators controlled by portion B will therefore alter the average displacement position of the combination of the two actuators. In practice, portion B of the control system varies the average

15   position of the actual actuators used by the vehicle.

Portion B of the control system outputs a required displacement demand, whilst portion A of the control system outputs a required velocity demand. Therefore, it is necessary that one of the outputs is

20   converted in the block 50, so that the outputs can be combined and used to control the actuators of the vehicle.

The present invention provides an active suspension control system which both controls the

25   actuators so that road inputs are not transmitted to the vehicle and also controls the actuators so that the average displacement of each actuator can be varied to provide a better vehicle ride and performance. For instance, the control system of the

30   invention a vehicle body can be prevented to a large extent from rolling as the vehicle corners. The portion B of the control system of the invention acts to prevent the actuators on the side of the car on the outside of a corner from deflecting in response to the

35 ·   roll forces on the vehicle caused by its cornering. In fact, to compensate for any flexibility in the

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suspension system the actuators can be extended on
roll.

     Since the microprocessor splits the input
signals from the sensors according to whether they
5   represent dynamic (ie generally relatively high
frequency) force inputs or steady state (ie generally
relatively low frequency) force inputs, the processing
of the signals can be achieved rapidly to produce
combined output signals. This permits the control
10   system of the invention to sample data at a fast rate
comparable with the rate of change of the various
signal values characteristic of vehicle active
suspension systems, thereby permitting accurate
control of the vehicle suspension system.
15   The applicant has envisaged a system wherein the
portion A of the control system samples at a higher
rate than portion B, this being advantageous in that
the number of calculations required by the control
system for a fixed time period is decreased. The
20   sampling rate for portion A must be reasonably high
since the said inputs will be of high frequency.
However, the sampling rate of portion B need not be so
high since the frequencies of the steady state signals
are generally far smaller.

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CLAIMS

1.   A vehicle suspension control system comprising;
        means for determining forces acting between the
sprung mass of said vehicle and one or more unsprung
masses connected thereto and producing signals
proportional to said forces;
        means for resolving said force signals to
represent steady state and dynamic force components
respectively;
        first means for determining, from the steady
state force signals, values of vector quantities
required between the sprung mass and the or each
unsprung mass to maintain a constant orientation of
said sprung mass relative to a reference plane;
        second means for determining, from the dynamic
force signals, values of vector quantities required
between the sprung mass and the or each unsprung mass
substantially to eliminate the transmission of dynamic
forces between the sprung mass and the or each
unsprung mass; and
        means for applying said required vector
quantities.

2.   A control system according to Claim 1 wherein
the means for detecting forces comprises sensor means
operable to produce a plurality of output signals, a
first set of said output signals corresponding
substantially to the dynamic force components and a
further set of said output signals corresponding
substantially to the steady state force components.

3.   A control system according to Claim 2 wherein
the first set of output signals comprises signals
representing the load measured in a load path between

an unsprung mass and the sprung mass.

4.   A control system according to Claim 2 or Claim 3
wherein the first set of signals comprises signals
5   representing the vertical accelerations of the
unsprung mass.

5.   A control system according to any of Claims 2 to
4 wherein the second set of signals comprises signals
10   representing one or more of:
        the displacement of an unsprung mass relative to
the sprung mass;
        the longitudinal acceleration of the vehicle;
        the lateral acceleration of the vehicle;
15   the rotational velocity of the vehicle;
        the speed of the vehicle; and
        the angle at which the vehicle is steered.

6.   A control system according to any preceding
20   claim wherein the signals representing the steady
state and/or dynamic force components are respectively
further resolved to represent a plurality of modal
force components acting on the vehicle.

25   7.   A control system according to any preceding
claim further comprising high pass filter means to
filter out low frequency components of said dynamic
force components.

30   8.   A control system according to any preceding
claim further comprising low pass filter means to
filter out high frequency components of said steady
state force components.

35   9.   A control system according to any preceding
claim further comprising simulator means for

simulating the ideal performance of the means for
applying the required vector quantities, and
comparator means for comparing the required vector
quantities with said ideal performance thereby to
5   produce a difference signal in response to which said
means for applying the required vector quantities
operates.

10.   A control system according to Claim 9 wherein
10   the difference signal is combined with an output
signal of the first means for determining values of
vector quantites as an input signal to the means for
applying said required vector quantities.

15   11.   A control system according to any preceding
claim wherein said first and second means for
determining values of vector quantities are capable of
operating substantially simultaneously.

20   12.   A control system according to any preceding
claim wherein the first and second means for
determining the values of vector quantities are
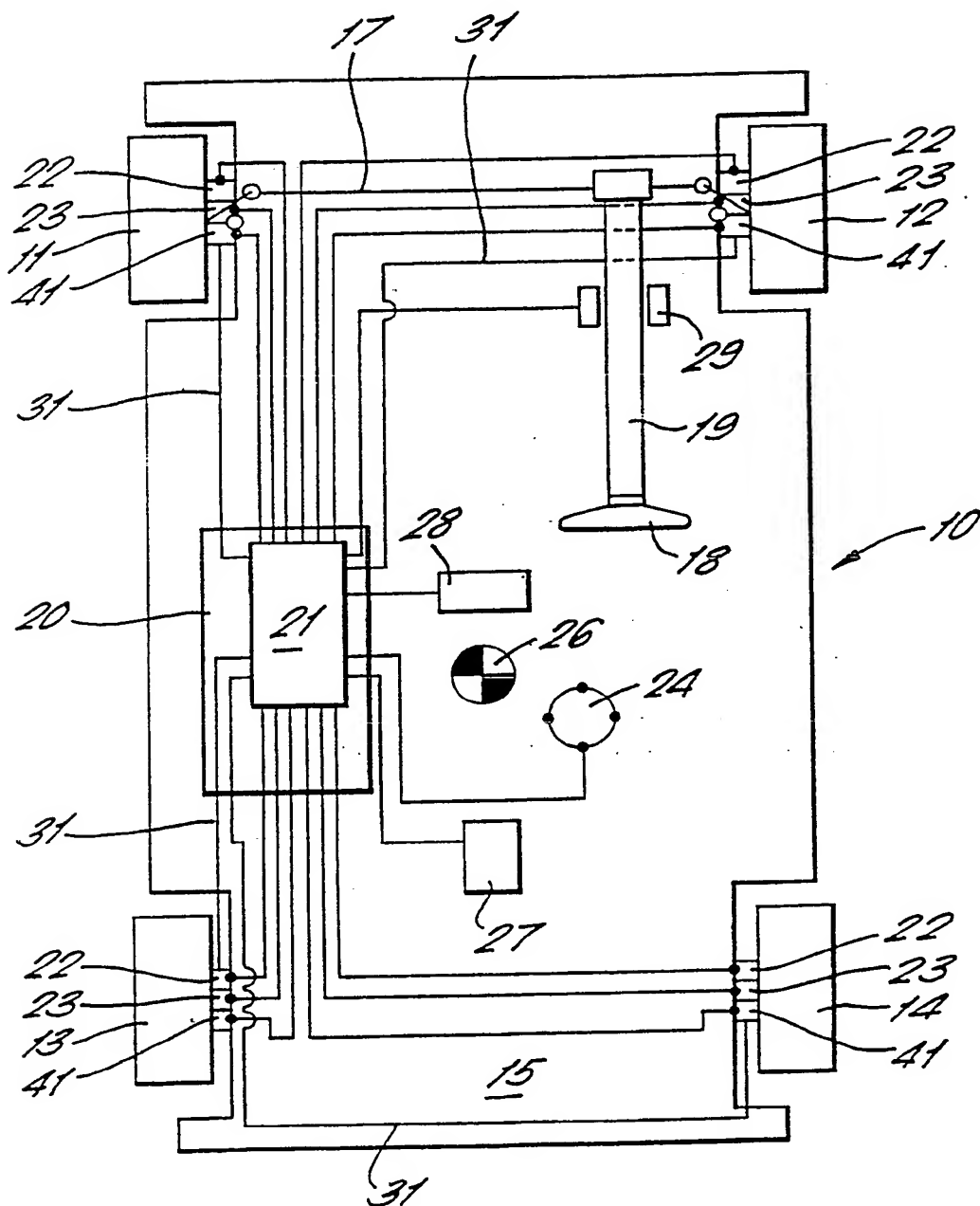constituted as a microprocessor.

25   13.   A control system substantially as herein
described, with reference to and as illustrated in the
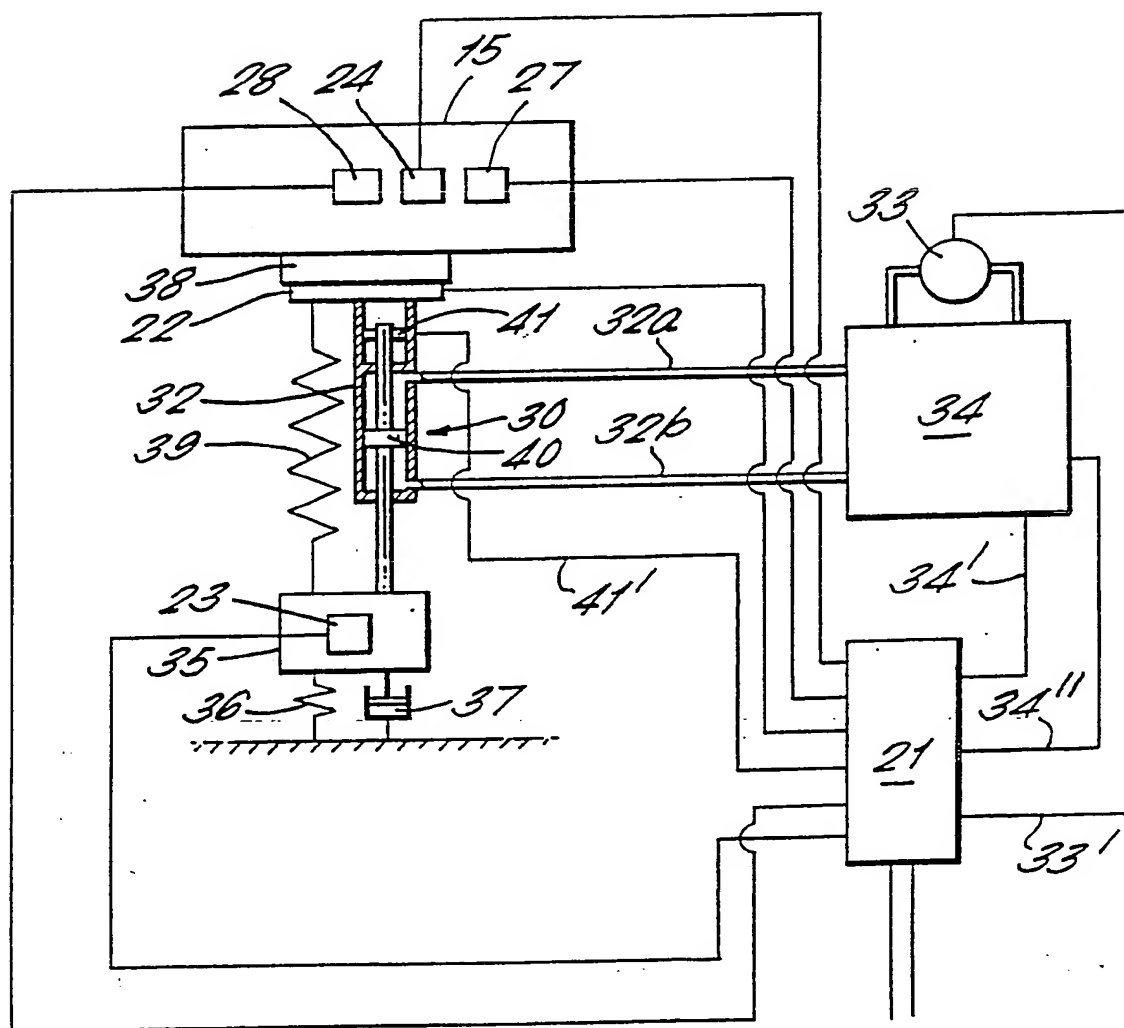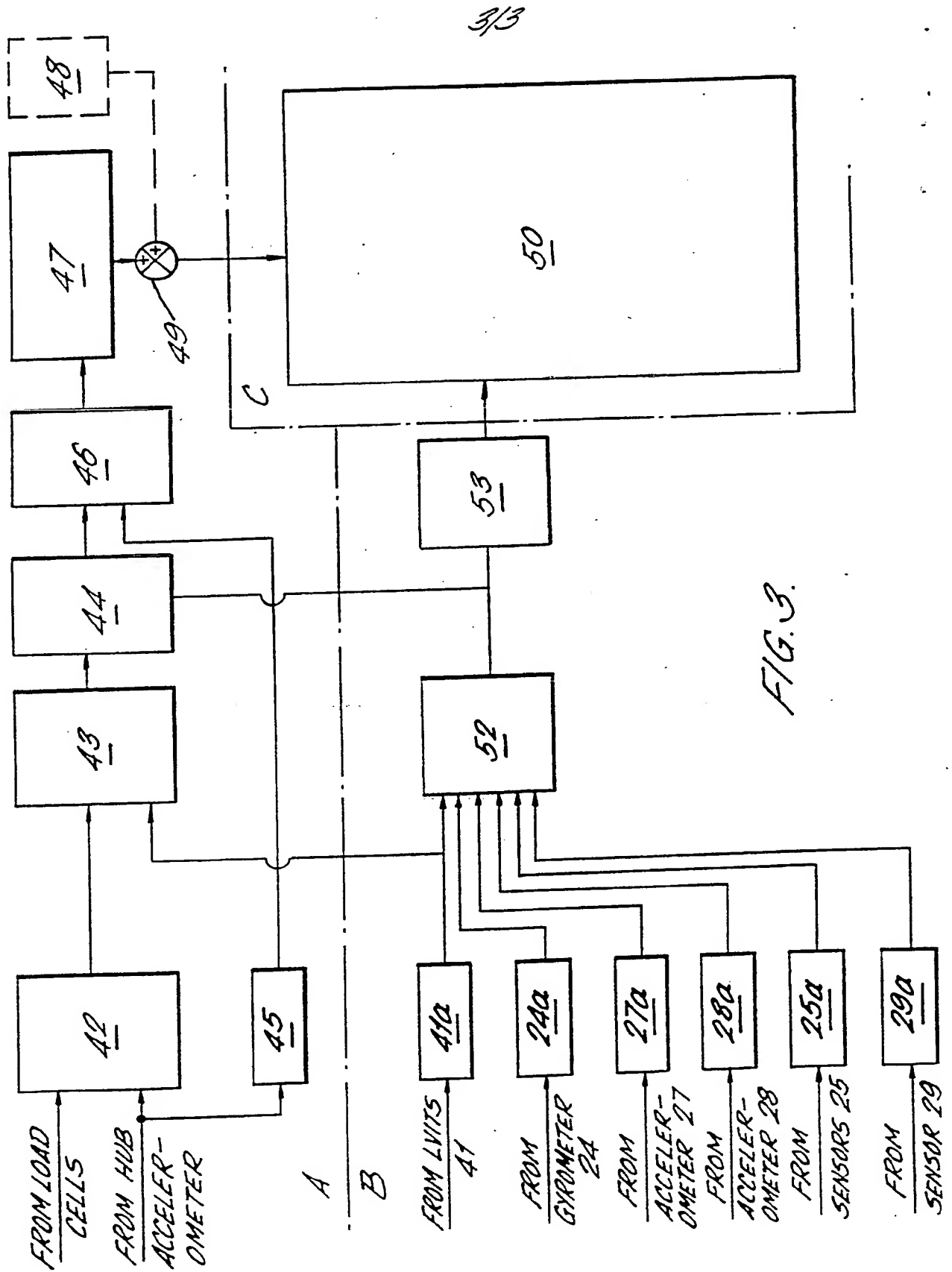drawings.

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FIG. 1.

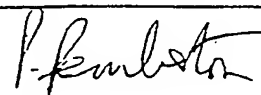FIG. 2.

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FIG. 3.

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C

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41a

24a

27a

28a

25a

29a

A

B

FROM LOAD CELLS

FROM HUB ACCELER-OMETER

FROM LVITS 41

FROM GYROMETER 24

FROM ACCELER-OMETER 27

FROM ACCELER-OMETER 28

FROM SENSORS 25

FROM SENSOR 29

I. CLASSIFICATION OF SUBJECT MATTER   (if several classification symbols apply, indicate all)[6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int.Cl. 5   B60G17/01

II. FIELDS SEARCHED

Minimum Documentation Searched[7]	
Classification System	Classification Symbols
Int.Cl. 5	B60G

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched[8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT[9]

Category[o]	Citation of Document,[11] with indication, where appropriate, of the relevant passages[12]	Relevant to Claim No.[13]
A	EP,A,249227 (NISSAN MOTOR CO LTD) 16 December 1987 see column 3, line 34 - column 6, line 17 see column 11, lines 15 - 36	1, 4, 11
A	PATENT ABSTRACTS OF JAPAN vol. 9, no. 183 (M-400)(1906) 30 July 1985, & JP-A-60 50013 (NISSAN JIDOSHA KK) 19 March 1985, see the whole document	1-3, 7, 8
A	EP,A,114757 (GROUP LOTUS CAR COMPANIES PLC) 01 August 1984 see the whole document (cited in the application)	1, 3-6, 12

-/--

° Special categories of cited documents :[10]

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
13 AUGUST 1990	- 7. 09. 90
International Searching Authority	Signature of Authorized Officer
EUROPEAN PATENT OFFICE	PEMBERTON P.

Form PCT/ISA/210 (second sheet) (January 1985)

| III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)		
Category °	Citation of Document, with indication, where appropriate, of the relevant passage	Relevant to Claim No.
A	EP,A,151421 (NISSAN MOTOR CO LTD) 14 August 1985 see page 2, line 35 - page 3, line 27; figure 2 see page 7, line 27 - page 8, line 34 see page 30, line 9 - page 31, line 24 ---	1, 3, 5, 7, 8
A	PATENT ABSTRACTS OF JAPAN vol. 12, no. 81 (M-676)(2928) 15 March 1988, & JP-A-62 221906 (NISSAN MOTOR CO LTD) 30 September 1987, see the whole document ---	1

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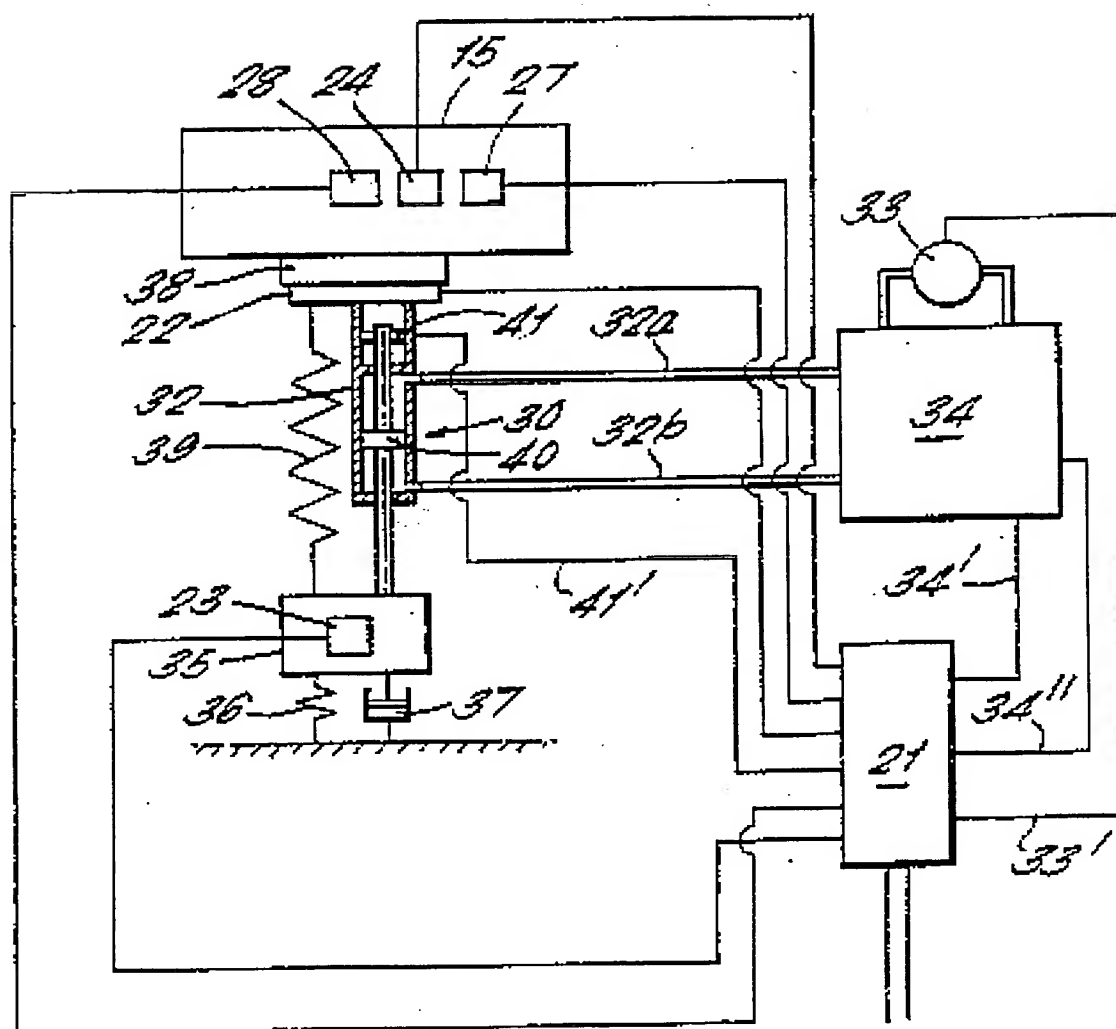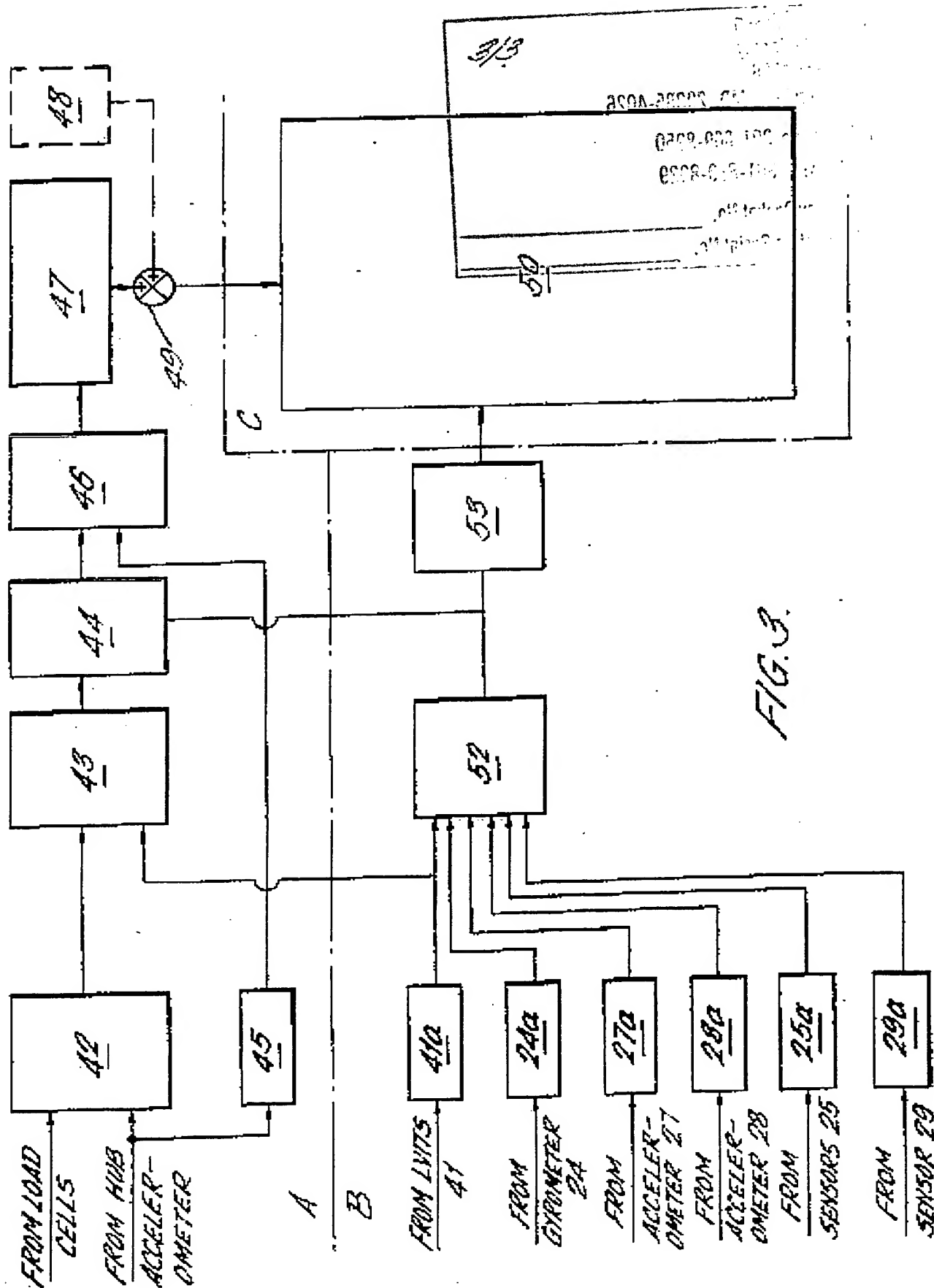| Patent document cited in search report	Publication date	Patent family member(s)	Publication date
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FIG. 1
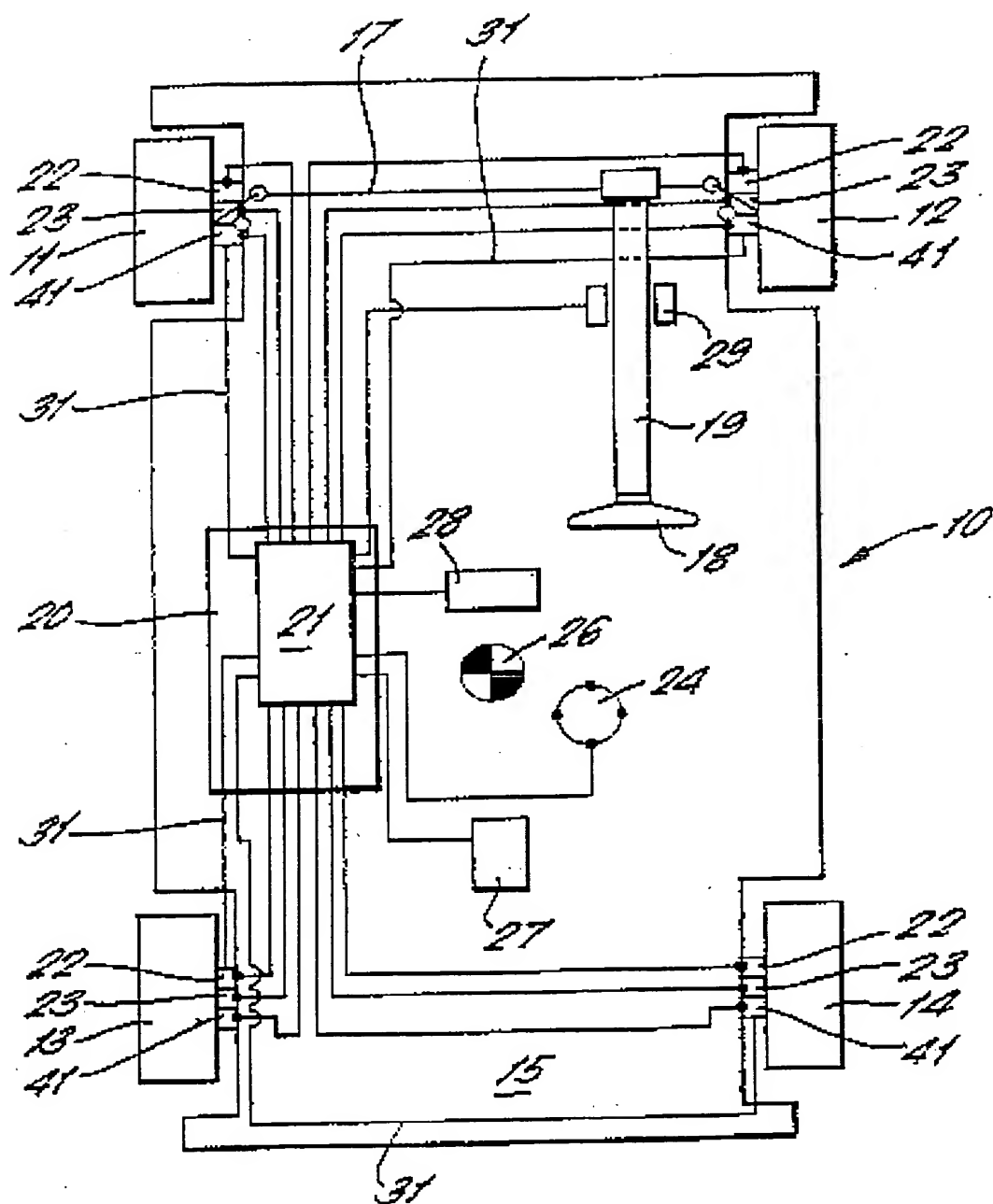
FIG. 2.

FIG.3.

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